PATENT SPECIFICATION

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COMPLETE SPECIFICATION

Multi-Stage Centrifugal Compressors

We, SULZER FRÈRES, SOCIÉTÉ ANONYME, a Company organised under the Laws of Switzerland, of Winterthur, Switzerland, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to multi-stage centrifugal compressors of single casing design, that is to say, compressors in which all the stages are enclosed in a single casing.

According to the present invention, in such a compressor the first stage comprises an open-sided rotor, or the first stage and at least one of its immediately succeeding stages comprise open-sided rotors, and all the stages subsequent to the said stage or stages comprise shrouded rotors.

An open-sided rotor has blades secured to one side of a rotating disc or formed integrally with the disc. The blade channels are bounded on one side by the said disc and on the other side by a stationary wall of the casing. Such rotors can run at high peripheral speeds and therefore produce a considerable rise in pressure. Their maximum flow capacity depends less on the inlet cross-section, which can be varied within wide limits, than on the outlet cross-section, since for a given peripheral velocity the width of the blade must not exceed a certain magnitude because of the risk of vibration effects. A disadvantage of open-sided rotors when used in multi-stage centrifugal compressors is that such rotors possess a very small working range, since if the quantity of the medium flowing through the compressor is slightly increased the outlet pressure falls considerably, while if the quantity of the medium flowing through is slightly reduced the surge limit is soon reached. Furthermore, open-sided rotors are very sensitive to axial displacements which may arise through differential heat expansion. Thus, the blades may either graze against the fixed wall of the casing bounding the blade channels, or the gap between this wall and the blades may become so large that losses through the gap substantially reduces the efficiency. The outside diameter of the rotors would also have to be considerably reduced in the later stages of a multistage compressor, because if the diameter remained unchanged the width of the blades of the rotors of the later stages would have to be made very small, owing to the reduction in volume of the medium during its compression in the preceding stages, and with such narrow blades the gap losses would become excessive. Thus the maximum possible pressure increase cannot be exploited with open-sided rotors.

For these reasons, in multi-stage centrifugal compressors all the stages generally comprise shrouded rotors. Here again the blades on one side are secured to, or are integral with, a rotating disc which also forms one boundary on the blade channel, but the other side of the blade channel is bounded by a shroud disc which is also secured to, or is integral with, the blades. Hence, no gap losses can occur from the blade channels. On its inner circumference the shroud disc generally is provided with a thick rib to give it the strength necessary to resist circumferential stresses due to centrifugal forces. Between this rib and the rotor shaft there is an annular inlet through which the medium enters the rotor. The maximum permissible peripheral velocity of a shrouded rotor is less than that of an open-sided rotor because the tension in the aforesaid rib arising from centrifugal forces must not exceed a limiting value. Since the blades of shrouded rotors mostly have a backward curvature, the bending stresses of these blades are increased as the peripheral velocity increases, which must also be taken into account. Thus for the same running speed and for optimum exploitation, a shrouded rotor must have a smaller outside diameter than an open-sided rotor. The maximum flow capacity of a shrouded rotor may depend partly on the outlet cross-section, as the bending stresses place a limit on the permissible thinness of the blades, but it depends chiefly on the cross-section of the annular inlet between the rib of the shroud disc and the rotor shaft. This cross-section cannot be arbitrarily increased because the diameter of the shaft cannot be

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reduced below certain limits, in view of the corresponding reduction of the critical speed, and because an increase in the diameter of the rib would further increase the tension in it and, owing to the restriction in the outside diameter, would reduce the length of the blades so that the desired increase of pressure could not be obtained.

As the flow capacity of a shrouded rotor is limited, so the quantity of the medium flowing through a multi-stage centrifugal compressor with shrouded rotors depends on the flow capacity of the first stage, and various attempts have been made to find a way of increasing the flow capacity of the first stage. A larger rotor could be used, or two rotors could be connected in parallel, or a low pressure rotor and a high pressure rotor arranged to rotate at different speeds could be connected in series. However, these solutions to the problem not only take up space but are also costly. It is already known to design at least the first stage of a multi-stage centrifugal compressor with a double-sided shrouded rotor, but while in this way a larger rate of flow is achieved the doubling of the first stage correspondingly increases the flow resistance, the space requirements and the cost. If, for example, the flow quantity is to be doubled, at least the first two stages must be provided with double sided shrouded rotors so that the medium can be compressed to an extent corresponding to the flow capacity of the single sided shrouded rotor of the next stage.

The present invention makes it possible to increase the quantity of flow while avoiding the aforesaid disadvantages.

If, for the sake of simplicity, one considers a particular multi-stage centrifugal compressor with shrouded rotors of which the first stage is so arranged that it causes an appropriate increase in pressure and at the same time has the optimum flow capacity, then the quantity flowing through can be considerably increased if the preliminary stage with an open-sided rotor is provided in the same casing. This increase in the flow quantity is greater than would be possible if a double-sided shrouded rotor were used instead. The maximum flow capacity of an open-sided rotor can be a multiple of the flow capacity of a shrouded impeller of the same diameter, so it is possible without difficulty to design the open-sided rotor in such a manner that the medium leaving it has a volume exactly corresponding to the flow capacity of the next stage employing a shrouded rotor of the same diameter.

If no additional increase in pressure is demanded from the compressor besides the increase of the quantity flowing through, then at least the last stage of the compressor can be omitted when the open-sided rotor is provided as the first stage. Since on the one hand an open-sided rotor causes a greater increase of pressure than a shrouded rotor of the same diameter, and since on the other hand the last stages of a compressor frequently have rotors of smaller diameter than the first stage (since otherwise the blade width would become too small owing to the reduction of the required outlet cross-section resulting from the compression of the medium), and therefore produce a smaller increase of pressure, it is possible in many cases when providing an open-sided rotor as the first stage to omit the last two stages employing shrouded rotors so that the whole compressor becomes smaller. Owing to the increased flow, what are now the last stages can be provided with blades of adequate width, even though the outside diameter of such rotors is not reduced. Thus all the rotors of the compressor can have the same outside diameter, which is a considerable simplification

It is advisable to provide a thrust bearing for the rotor shaft close to the open-sided rotor of the first stage. Then in the event of unequal thermal expansion occurring, the first stage suffers the minimum axial displacement so that the aforementioned sensitiveness of an open-sided rotor to unequal thermal expansions is guarded against. In any case, the blades are prevented from grazing against the fixed wall of the casing which could lead to fracture of the blades, as in the proposed arrangement thermal expansion will generally have the effect of slightly increasing the gap between the blades and the wall of the casing.

The invention may be performed in various ways, and one particular four stage compressor embodying the invention will now be specifically described by way of example with reference to the accompanying drawing, which is a longitudinal section through the compressor.

In the compressor shown in the drawing the medium to be compressed enters the compressor through an inlet 2 formed as part of of the casing 1. The incoming medium is directed by guide vanes 3 into the blade channels 4 of the first compression stage, which comprises an open-sided rotor. The blade channels 4 of the first stage are bounded on one side by the rotor disc 5, with which the blades are connected, and on the other side by a stationary wall 6 fixed in the casing. Next, the medium enters a fixed diffuser section 7 in which part of its kinetic energy is converted into pressure energy. Then the medium is guided radially inwards through a fixed return channel 8 and enters the blade channel 9 of the second stage rotor. This rotor is a shrouded rotor, the blades being connected at one side to the rotor disc 10 and on the other to a shroud disc 11. The inside circumference of the shroud disc is strengthened by a thick rib 12. The medium emerging from the second rotor passes through a fixed diffuser section 13. The compression in the two subsequent stages is carried out in the same manner as in the second stage. The com-

pressed medium leaves the compressor through an outlet 14 which is also formed as part of the casing, which outlet may comprise an annular or volute chamber.

The rotor discs 5 and 10 are mounted on the rotor shaft 15 with spacers 16 between them. The shaft 15 is supported at one end in a journal bearing 17 which is itself mounted in a supporting casing 18, while at the other end the shaft is supported in a combined journal and thrust bearing 19 which is itself mounted in a supporting casing 20. The bearing 19 is arranged as close as conveniently possible to the open-sided rotor of the first compression stage. The shaft 15 is connected with its driving shaft through a coupling 22. The interior of the casing 1 is sealed from the outside by packings 23, while an additional packing 24 is provided to ensure additional security against leakage due to longitudinal expansion and contraction of the rotor shaft.

If the critical speed permits the employment of a longer rotor shaft, the compression could be carried out with five or more stages.

A slight reduction in the axial length can be achieved by omitting the guide vanes 3 and, instead, shaping the inlet ends of the blades in such a manner that they deflect the medium without shock into the blade channels 4 of the first stage.

The centrifugal compressor illustrated has rotors in which the medium is compressed by a movement which is essentially in the radial direction, but the invention is also applicable to compressors having rotors in which the blade channels also have an axial component i.e. rotors of somewhat conical form.

Should the amount of the medium delivered by a single open-sided rotor be too small, the second compression stage could also comprise an open-sided rotor. Compressors in which more then the first two stages comprise open-sided rotors would only be required in exceptional cases.

What we claim is:—

1. A multi-stage centrifugal compressor of single casing design in which the first stage comprises an open-sided rotor, or the first stage and at least one of its immediately succeeding stages comprise open-sided rotors, and all the stages subsequent to the said stage or stages comprise shrouded rotors.

2. A multi-stage centrifugal compressor as claimed in Claim 1 in which all the rotors have the same outside diameter.

3. A multi-stage centrifugal compressor as claimed in Claim 1 or Claim 2 in which a thrust bearing for the rotor shaft is provided close the open-sided rotor of the first stage.

4. A multi-stage centrifugal compressor substantially as specifically described with reference to the accompanying drawing.

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Agents for the Applicants.

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754,322 COMPLETE SPECIFICATION
1 SHEET
This drawing is a reproduction of the Original on a reduced scale.